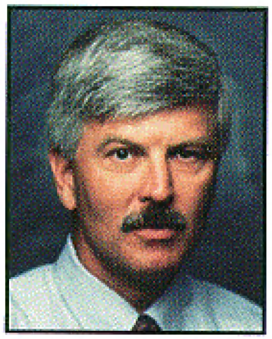
Case histories illustrate the importance of transient data analysis



by Richard Thomas, P.E.
Principal Engineer
Machinery Diagnostic Services
Bently Nevada Corporation
Baton Rouge, Louisiana

Vibration data can be acquired from a machine when it is operating at constant speed and load, and when speed or load are changing. Each is important for understanding the machine's condition. Steady state data, acquired under constant machine conditions, is an essential part of most machinery predictive maintenance programs. However, many of these programs neglect to obtain the important data available from startups and shutdowns: Transient data.

In the last issue of the Orbit, I described transient data, the plots used for its display, and certain machine characteristics that these plots show. Bently Nevada's Machinery Diagnostic Services (MDS) group makes extensive use of transient data when analyzing machine malfunctions. The case histories in this article show how transient data helped solve actual machine problems.

Excessive shaft average centerline position change

Operators at a power plant were concerned about excessive vibration in an 1150 MW steam turbine generator. The turbine and generator had been originally commissioned in 1985. In the first five years of its operation, maintenance was required mainly to balance the turbine and align the exciter. However, as the machine passed through a certain load range during startups and shutdowns, the HP turbine vibrated severely. Shaft average centerline data, captured during a startup, helped solve this problem.

The steam turbine had four stages, all of double flow design: One high pressure (HP) turbine and three low pressure (LP) condensing turbines. The steam turbine and generator were rigidly coupled. The generator consisted of a four-pole generator, operating at 1800 rpm, and an exciter. Each of the 11 bearings in the machine train had a pair of Dual Probe transducers installed in an XY configuration (Figure 1). Among the measurements a Dual Probe transducer makes is absolute rotor motion, that is, rotor motion relative to free space. It consists of two transducers: A velocity Seismoprobe® sensor that measures absolute casing or foundation vibration, and a proximity probe that measures rotor motion relative to the bearing housing. A Keyphasor® probe was installed on the turbine shaft to provide a phase reference.

Bently Nevada MDS analyzed this machine in October, 1990. Plant personnel told us that, during startup, the HP turbine would vibrate and shake its foundation as it passed through its balance resonance speed region. In a two hour period on the morning of 15 October, the turbine and generator were brought up to synchronous speed (1800 rpm). We used a stack of 108 Data Acquisition Instruments and an ADRE® 3 System to capture transient data from the

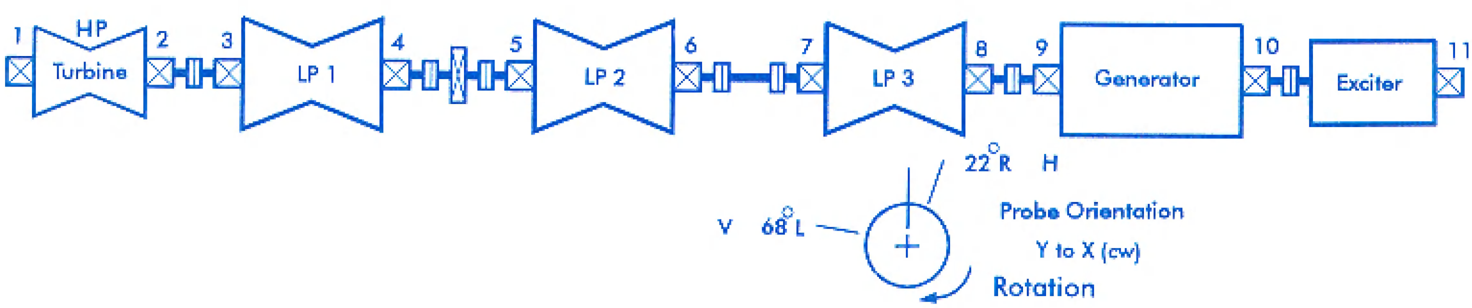


Figure 1
1150 MW steam turbine - generator machine train.

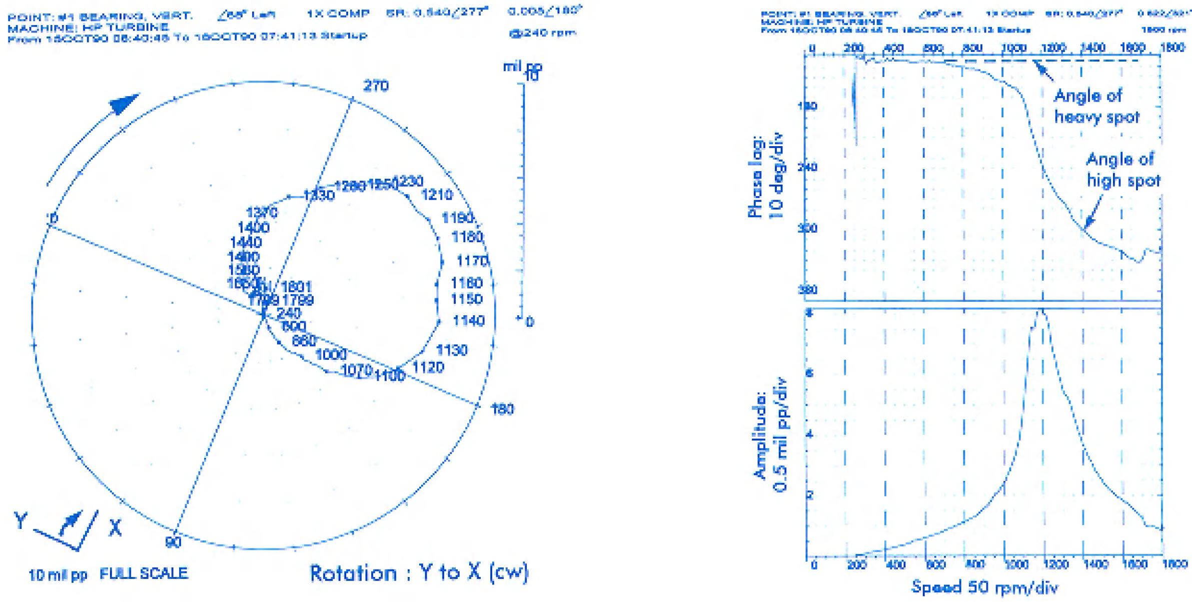


Figure 2
1X transient response at Bearing 1 vertical transducer, before bearing clearance was properly set. (a) polar (b) Bode

startup. From the 15th through the 18th of October, load was increased, to 1150 MW (full load).

We examined transient data from HP turbine Bearings 1 and 2 and found that:

- HP Turbine rotor balance is not a problem. No load, 1800 rpm, and full load 1X vibration amplitudes are 1 mil (25 μm) pp or less at both bearings (Figures 2a and b).
- The Synchronous Amplification Factor (Q_S) at Bearing 1 is greater than 7, which is unusually high (in Figures 2a and b, Q_S is calculated by the ratio method). Q_S helps us estimate the vibration amplitudes that occur at rotor natural frequencies due to the unbalance force. Q_S is approximately 10, which indicates that the system is susceptible to high vibration at resonance.
- In Figure 3 are two shaft average centerline plots. A shaft average centerline plot shows average rotor position versus time; on these plots, the dotted circle represents the bearings' 20 mil (500 μm) diametral clearance. Figure 3 shows that at operating speed, Bearing 2 had an eccentricity ratio of 0.4 and a rotor position angle of 120° (reference is vertical down). Therefore, the rotor's centerline, at Bearing 2 (and Bearing 1), is in the bearing's upper left quadrant. In a machine with fluid-film bearings, such as this, the rotor normally rises on an oil wedge as it gains speed. A shaft rotating Y to X (cw) with no abnormal load will usually be in the lower left quadrant of its bearing, with a rotor position angle between 20° and 50° (reference is vertical down).
- Figure 3 also shows that, as load (steam flow) increases, the rotor position at Bearings 1 and 2 changes dramatically. Shortly after 9 am on 15 October, the load is increased and at Bearing 1, the rotor lifts up and to the right, 11.5 mils (292 μm) right and *34 mils (864 µm) above* the zero speed reference position (the bottom of the bearing). This coincides with the 850 MW load point. As load ▶

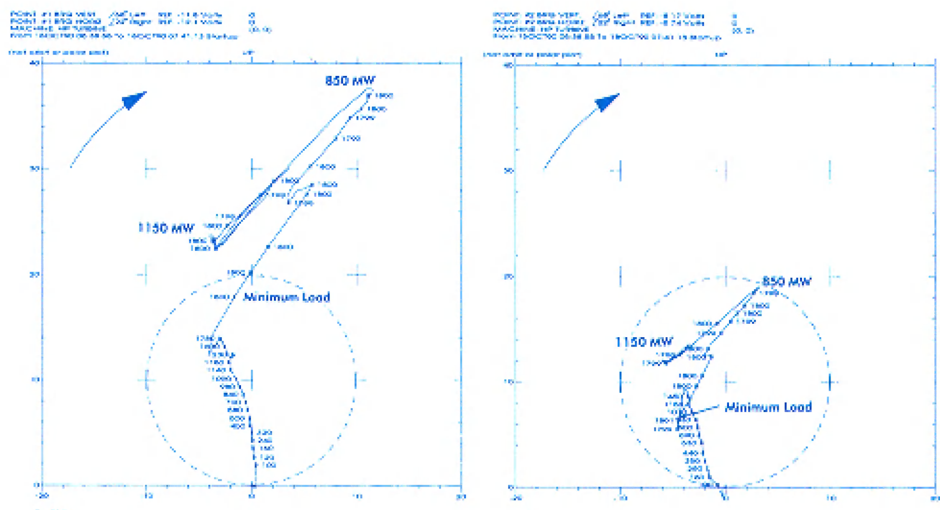


Figure 3
Shaft average centerline position changes at Bearings 1 and 2, before bearing clearance was properly set.

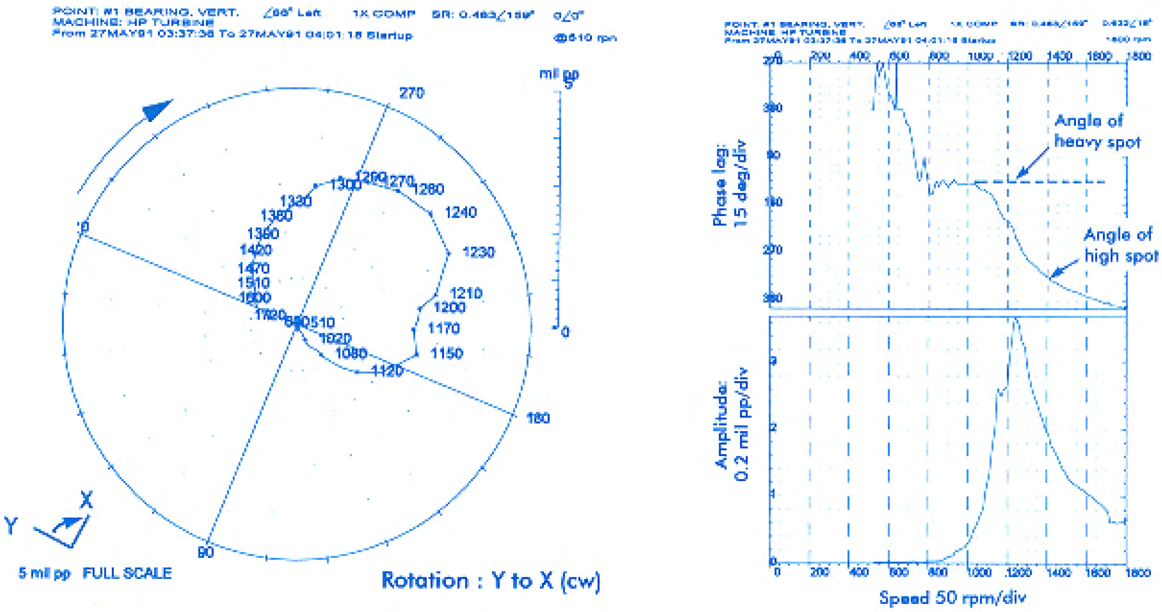


Figure 4
1X transient response at Bearing 1 vertical transducer, after bearing clearance was properly set. (a) polar (b) Bode

increases further, the rotor moves down and left, to a final position *4 mils (100 μm) left and 21 mils (530μm) above the reference position.*

We investigated the HP turbine's inlet steam valve sequencing. Steam entered the turbine through upper and lower valve racks. In the load range from 0 to 850 MW, the lower valve rack opens, admitting steam from the bottom, which lifts the rotor. Above 850 MW load, the upper valve rack begins to open and admit steam from above the rotor, which loads the rotor from above. In Figure 3, notice that the shaft moves in the same direction at both Bearings 1 and 2. However, at Bearing 2, the shaft does not move beyond the bearing clearance; at Bearing 1, the shaft moves outside the bearing clearance.

Because Bearing 1's Synchronous Amplification Factor was higher than normal, and the shaft centerline at Bearing 1 showed abnormal motion, we suspected that:

- The internal clearance at Bearing 1 was excessive.
- Bearing 1 was loose in its pedestal, and, therefore, was not properly restrained.
- A combination of both.

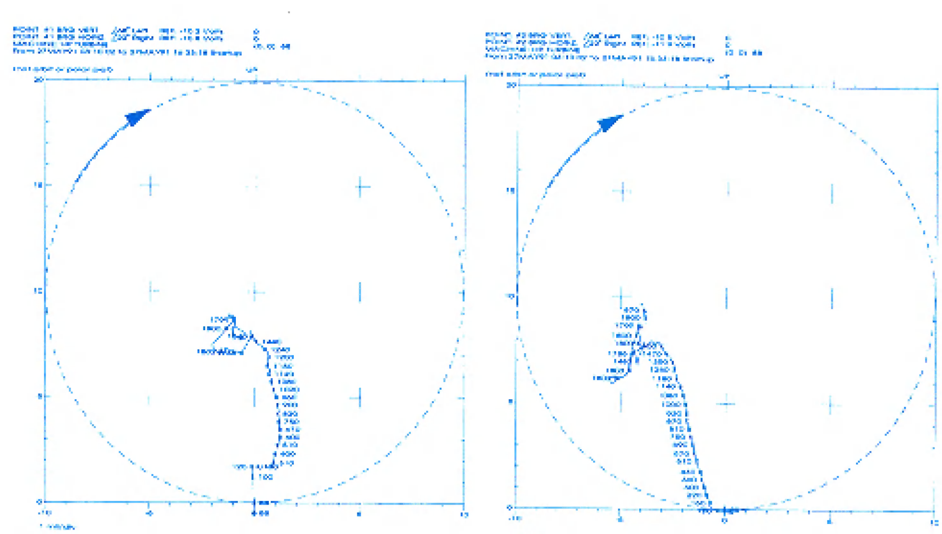


Figure 5
Shaft average centerline position changes at Bearings 1 and 2, after bearing clearance was properly set.

Full load vibration levels were acceptable; therefore, operators decided to wait until the next scheduled shutdown to inspect Bearing 1.

When Bearing 1 was later inspected, plant maintenance personnel found excessive clearance. Bearing 1 was a four pad, tilt pad bearing in which all four pads could be shimmed. To properly set the bearing's internal clearance, the two upper pads required shimming. They were not shimmed; it had been overlooked during final assembly in 1985.

After Bearing 1's clearance was properly set, the machine was restarted in May 1991. A polar and a Bode plot, generated from startup data (Figures 4a and b) showed that, as the rotor passed through its critical speed (1200 rpm), maximum 1X amplitudes did not exceed 4 mils (100 μm) pp. Shaft average centerline plots generated from the same

data show that the shaft average centerline, at both Bearings 1 and 2, remained in the lower left quadrant of each plot throughout the entire speed and load range (Figure 5).

Excessive clearance at Bearing 1 had reduced the quadrature dynamic stiffness of the rotor/bearing system. At resonance, quadrature dynamic stiffness limits rotor response — it restrains rotor motion. With less effective damping at Bearing 1, rotor vibration at resonance was greater than it otherwise would have been. The heavy rotor was well balanced, so vibration above resonance was acceptable.

It is interesting to compare polar and Bode plots generated from data acquired before and after Bearing 1 was shimmed (Figures 2 and 4). Not only is it obvious that vibration amplitude has decreased, but the resonance frequency has increased. When Bearing 1 was loose in its pedestal, the resonance frequency was 1190 rpm. After Bearing 1 was properly shimmed, the mechanical stiffness increased, as is evidenced by an upward shift in the resonance frequency to 1240 rpm.

The shaft average centerline plot can be very useful in certain circumstances. In this instance, its two-dimensional representation of the average position of the shaft within its bearing made it very clear that this machine's problem was at Bearing 1.

Q_S, the Synchronous Amplification Factor

The Synchronous Amplification Factor, Q_S, is a measure of the amount of synchronous quadrature dynamic stiffness present in a rotor bearing system at a mechanical resonance. At a mechanical resonance, synchronous quadrature dynamic stiffness is the rotor bearing system's only restraint. High Q_s values indicate that the synchronous quadrature dynamic stiffness at resonance has decreased to the point where vibration amplitudes at resonance are potentially damaging to the machine.

To calculate Q_S by the ratio method, divide the vibration amplitude at resonance by the vibration amplitude at high rotative speed (but below the next higher mode range).

Average eccentricity ratio and rotor position angle

The average eccentricity ratio is a relative measure of the shaft's position between the center of the bearing and the bearing wall. It is calculated by dividing the average position of the shaft centerline, relative to the bearing center, by the bearing (or seal) radial clearance.

The rotor position angle is the angle between an arbitrary reference through the center of a bearing (usually vertical down) and a line connecting the bearing and shaft centers, measured in the direction of rotation. ■

Higher-order transient data helps identify a shaft crack

A nuclear power plant had serious shaft cracks in its main coolant pumps in 1989, 1990 and 1991. Bently Nevada MDS identified these shaft cracks after reviewing several types of machine data. Some of the data we used was transient data from the fifth harmonic of running speed.

It takes a large amount of varied data to accurately determine if a machine has a shaft crack. Transient data is important, but is not the only data required. Among the data required:

- Historical trends of overall vibration amplitude.
- 1X Acceptance Region data and slow roll data. If a rotor is cracked, it will bow and modify the 1X response.
- 2X Acceptance Region data. As a transverse crack grows, shaft stiffness becomes asymmetric in the plane of the crack. A rotor with shaft asymmetry will, *in the presence of a significant radial load*, generate a 2X response.
- Natural frequencies, and any changes in those frequencies, as shown in transient data plots. A transverse crack reduces shaft stiffness, which lowers its natural frequencies. For a given rotor mode, the closer a crack is to the rotor's point of maximum deflection, the more the crack will lower the natural frequency.
- A good rotor model. It is an aid in evaluating rotor mode shapes and in predicting how a crack will change the natural frequencies.

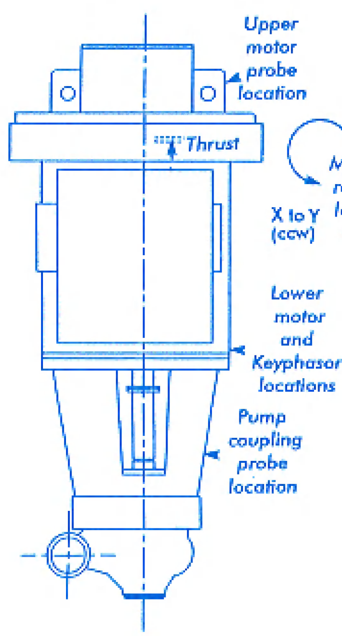


Figure 6
Main coolant pump.

Because this article focuses on transient data analysis, I will only discuss how the shaft cracks were evident in higher-order, 5X polar and Bode plots. The data presented here is from cracked shafts that occurred in 1989 and 1990.

The main coolant pumps were large vertical pumps driven by 7,940 hp electric motors (Figure 6). Normal pump speed was 1790 rpm, and nominal flow rate was 150,000 to 170,000 liters per minute (40,000 to 45,000 gallons per minute). The pump impellers had 5 vanes.

In analyzing a machine for the presence of a cracked shaft, 1X transient response can indicate changes in balance resonances that occur within the normal operating speed of the rotor. The first balance resonance frequencies of these pumps were slightly above 1800

rpm, primarily due to the electric motor's translational response. In this mode, maximum deflection occurred at the motor rotor's mid-span, while little 1X vibration occurred from the coupling to the pump impeller. The second resonance frequency of these pumps was from 3100 to 3200 rpm, and is the response of the pump shaft. Maximum deflection occurs at the pump impeller, very little occurred from the top of the motor to the pump coupling.

The mode shapes of these two resonances suggested that a transverse crack on the pump rotor would have little effect on the first balance resonance, but a significant effect on the frequency of the second resonance. A principle of higher order transient analysis is that a rotating shaft can create forcing functions at integer multiples of rotative speed. Therefore, on this machine, as shaft rotative speed passes through 1550 rpm, a 2X forcing function passes through 3100 cpm. Remember that, on these pumps, the point of maximum deflection for the second resonance occurs at the impeller. The impeller's 5 vanes interact with the system hydraulics to create a natural forcing function at 5 times shaft rotative speed. If we tracked 5X impeller blade pass frequency from 330 rpm to 1,800 rpm, we could observe the response of the pump rotor from 1650 rpm to 9000 cpm.

A pump in good condition had a second resonance frequency between 3100 and 3200 cpm. Figures 7a and b show this, in 5X response polar and Bode plots from a typical pump. The same plot types, from the 1989 cracked pump shaft, show that the second resonance peak shifted down to 2540 cpm, with 5X vibration amplitude greater than 5 mils (125 μm) pp (Figures 8a and b). Likewise, the 1990 cracked pump shaft lowered the second resonance frequency to 2955 cpm (Figures 9a and b).

While this was not the only data we used to determine that the shafts were cracked, it should be clear that these higher-order, 5X response plots were very important.

ADRE® for Windows

All transient data shown in this article was captured and processed with Bently Nevada portable diagnostic instruments: The ADRE 3 System, and our new ADRE for Windows System. All plots for this article were generated by ADRE for Windows, which replaces our ADRE 3 System. Each system can capture and process up to 16 channels of vibration and process data.

ADRE for Windows Software is powerful, yet easy to use. It runs under Microsoft Windows on PC compatible, 386 or better computer systems (a notebook computer is best for this portable system). You can view data as it is collected, store it for later analysis, or both. You can view data from several transducers, each in its own window, view data from one transducer in several different plot formats, each in its own window, or both. You can leave the system in place, ready to begin collecting data if a change occurs in vibration amplitude or phase, a process transducer level, or other events. ADRE for Windows graphs data in many powerful formats: Bode, polar, orbit, timebase, spectrum, cascade, waterfall, shaft average centerline, trend, rpm versus time, and others. When you've finished, you can import ADRE for Windows data and plots into a spreadsheet or word processing program, for easy report writing. ADRE for Windows is the most advanced portable diagnostic system available.

Conclusion

While most machinery predictive maintenance programs use steady state data, many don't use transient data. This is unfortunate, because transient data is often essential for accurate machinery audits. I hope that the case histories described here have convinced you of the value of transient data analysis.■

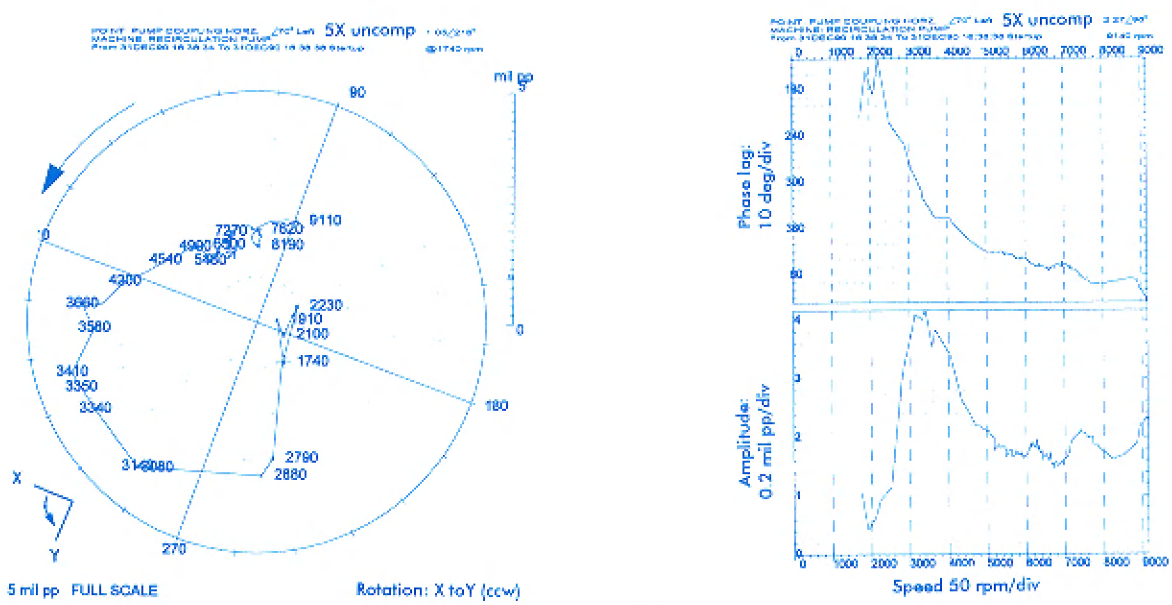


Figure 7
5X transient resonance of typical coolant pump with no shaft crack present. The second resonance frequency occurs at 3140 cpm. (a) polar (b) Bode

Acceptance Region plots

An Acceptance Region plot is a trend plot, of 1X or 2X vibration vectors (amplitude and phase lag angle) presented in polar format, or shaft average centerline position in XY (rectangular) format. The user defines the normal Acceptance Region for each shaft radial vibration or position measurement on the machine, based on historical data for the machine under all normal operating conditions. Acceptance Region information is an important indicator of a shaft crack. ■

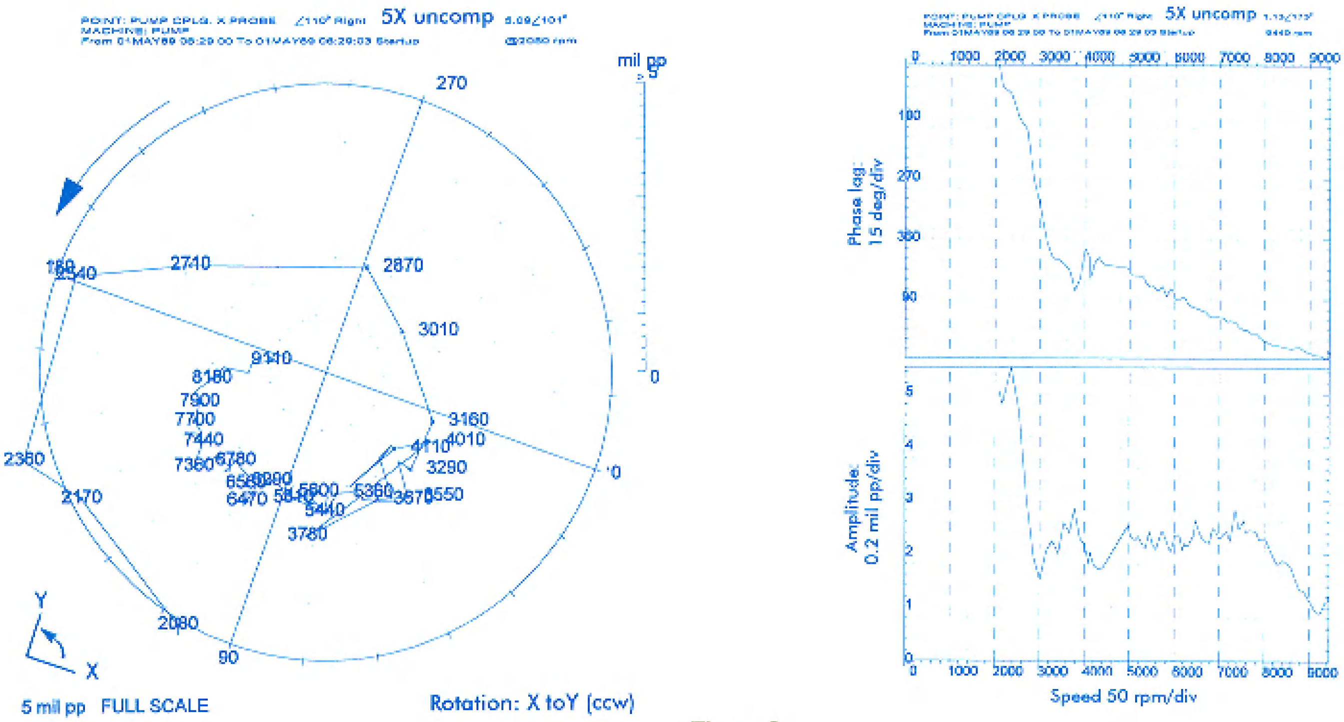


Figure 8
5X transient resonance of coolant pump with a cracked shaft (1989).
The second resonance frequency occurs at 2540 cpm. (a) polar (b) Bode

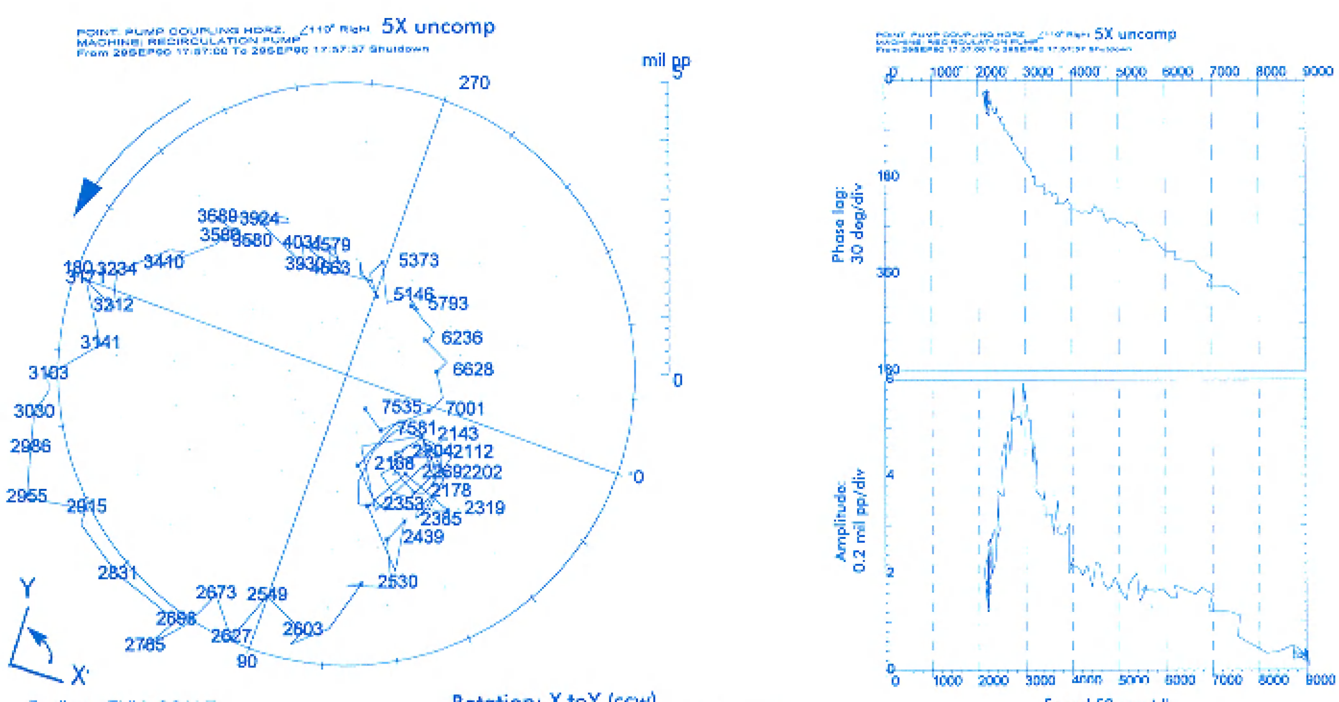


Figure 9
5X transient resonance of coolant pump with a cracked shaft (1990).
The second resonance frequency occurs at 2955 cpm. (a) polar (b) Bode